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THESIS

HEAT TRANSFER STUDIES ON A RECTANGULAR
CHANNEL WITH OFFSET PLATE FINS

by

Jeffrey M. Masterson

December, 1993

Thesis Advisor:

Y.K. Joshi

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CHANNEL WITH OFFSET PLATE FINS

by

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Lieutenant, United States Navy
B.S., United States Naval Academy, 1986

Submitted in partial fulfillment
of the requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

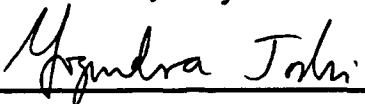
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ABSTRACT

Convective heat transfer characteristics of a liquid cooled rectangular channel, containing offset plate fins were investigated experimentally. The selected geometry was a 10x model of the fluid circulation passages found in the commercially available SEM-E type electronics cooling module. The test surface containing fins was made of aluminum and heated at its base by a thermofoil heater. The Reynolds numbers were varied between 100-800, with water as the cooling fluid. Surface temperature measurements on the heated surface were used to determine the Colburn j factor. The effects of natural convection were also investigated.

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NOMENCLATURE

A_c	-channel free flow area (179.2 mm ²)
A_t	-total test section free flow area (3506 mm ²)
A_h	-test section heated area (0.1116 m ²)
A/l	-heat transfer area per unit fin length (0.04370 mm)
D_h	-hydraulic diameter (0.01641 m)
f	-Fanning friction factor
h	-fin height (15.24 mm)
I_h	-heater current (A)
j	-Colburn j factor
k_f	-fluid thermal conductivity (W/m-K)
l	-fin length (31.75 mm)
Nu	-Nusselt number
Pwr	-heater power (W)
Pr	-Prandtl number
q''	-heat flux (W/m ²)
Q	-volumetric flow rate (m ³ /s)
Re	-Reynolds number
s	-fin spacing (11.76 mm)
t	-fin thickness (1.52 mm)
T_{AVG}	-average test section temperature (K)
T_{IN}	-test section inlet temperature (K)
T_{PROP}	-fluid property determination temperature (K)
v	-mean fluid velocity (m/s)

V_h -heater voltage (V)
 V_{pr} -precision resistor voltage (V)
 ρ -fluid density (kg/m³)
 μ -fluid viscosity (m²/s)

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I. INTRODUCTION

A. ELECTRONIC COOLING

With the advent of the transistor, designers have been faced with increasingly difficult challenges in the area of electronic cooling. Component miniaturization has progressed faster than reductions in electronic power generation. For reliable operation, the chip junction temperature typically must remain below 85°C. This requirement forces designers to use more aggressive methods of heat removal as chip power densities increase.

Many options are being investigated to provide increased heat removal capability, including conduction cooling and direct immersion cooling. This study focuses on flow through liquid cooling in offset plate fin heat exchangers.

1. Offset Plate Fins

Offset plate fins have been used in heat exchange equipment for many years. Their large surface area to volume ratio leads to increased heat transfer. This enhancement comes from the periodic interruption of the thermal boundary layer as the flow passes over the fins. The increase in heat transfer, however, is accompanied by an increase in pressure drop due to the friction and form drag of the fins.

A typical offset plate fin arrangement is shown in Figure 1. The fins are characterized by their length (l), height (h), thickness (t), and spacing (s). The offset is normally half the fin spacing which places them in the center of the channel formed by the previous row. The fin geometry is expected to have an effect on the heat transfer performance and the pressure drop of the offset plate fins. The exact relationship has not been determined but is currently being researched [Ref. 1].

The complex flow path around the fins has prevented the development of good predictive models or generalized correlations. This has forced researchers to use empirical data as the basis for their heat exchanger designs.

2. Offset Plate Fin Heat Exchangers

The typical design of the offset plate fin heat exchanger is shown in Figure 2. It consists of fabricated aluminum strip fins brazed in between aluminum plates. The fins are characterized by the same dimensions as the offset plate fins in Figure 1 with the only difference being the lateral ridge created during manufacturing. This loss of area must be taken into account for calculation of the hydraulic diameter, D_h .

One specific offset plate fin heat exchanger that is becoming widely used in electronics thermal control applications is the SEM-E (Standard Electronic Module, Format

E). This is a standard military board that uses the typical design discussed above and shown in figure 2. A complete description of the SEM-E is contained in Ref. 1.

A rough sketch of a SEM-E is shown in Figure 3 [from Ref. 2]. Three passes are used to enhance the heat transfer through the core. The offset fins are only installed in the horizontal runs to prevent impeding the flow from pass to pass.

B. PREVIOUS RESEARCH ON OFFSET PLATE FIN HEAT EXCHANGERS

1. Air Cooled Modules

Offset plate finned surfaces using air have been studied since the 1940's. One of the most comprehensive compilations of available data is by Manglik and Bergles [Ref. 3]. It includes an annotated bibliography of previous research and summarizes existing empirical correlations. Using the data previously gathered and correlating it is probably the most realistic method to predict the performance of untested offset plate fin heat exchangers [Ref. 4]. Manglik and Bergles used this technique to correct previous data for consistency and developed improved correlations for the Colburn j factor and the Fanning friction factor, f .

Their work is assumed applicable for Prandtl numbers in the range of 0.5 to 15 so the correlations could be applied when using water in normal temperature ranges. However, the correlations would not be valid using a heat transfer oil with

a Prandtl number in the range of 40 to 100. The effects of these high Prandtl number coolants on the heat transfer performance of offset plate finned heat exchangers is as yet largely unknown.

2. Liquid Cooled Modules

The low heat transfer coefficients and heat capacities of air restrict it's use to relatively low power densities in electronics cooling. For higher power use, one option is flow through liquid cooling. Relatively little research has been done in this area. There is an experimental program under way at the CALCE Electronics Packaging Research Center at the University of Maryland to begin quantifying the effects of using high Prandtl number liquids as the coolant [Ref. 1].

Their study involves using water and PAO (polyalphaolefin, a synthetic oil) as the coolant through a variety of fin geometries. The key result of their effort so far has been to establish the clear Prandtl number dependence on the Colburn j factor. This factor can be increased without a significant accompanying increase in the Fanning friction factor by using high Prandtl number fluids. This result invalidates the theory that the correlations previously mentioned for air could be applied to liquid cooled modules.

C. OBJECTIVES OF PRESENT STUDY

This study experimentally investigated the capacity for heat removal of a scaled up model of the SEM-E using water as

a coolant. By varying the flow rate through the fins and the power input, j was determined over the range of interest.

Specific objectives were:

- To determine the variation of Colburn j factor with Reynolds number.
- To determine the effects of natural convection on the Colburn j factor.
- To detect the presence of temperature variations in individual rows of fins.

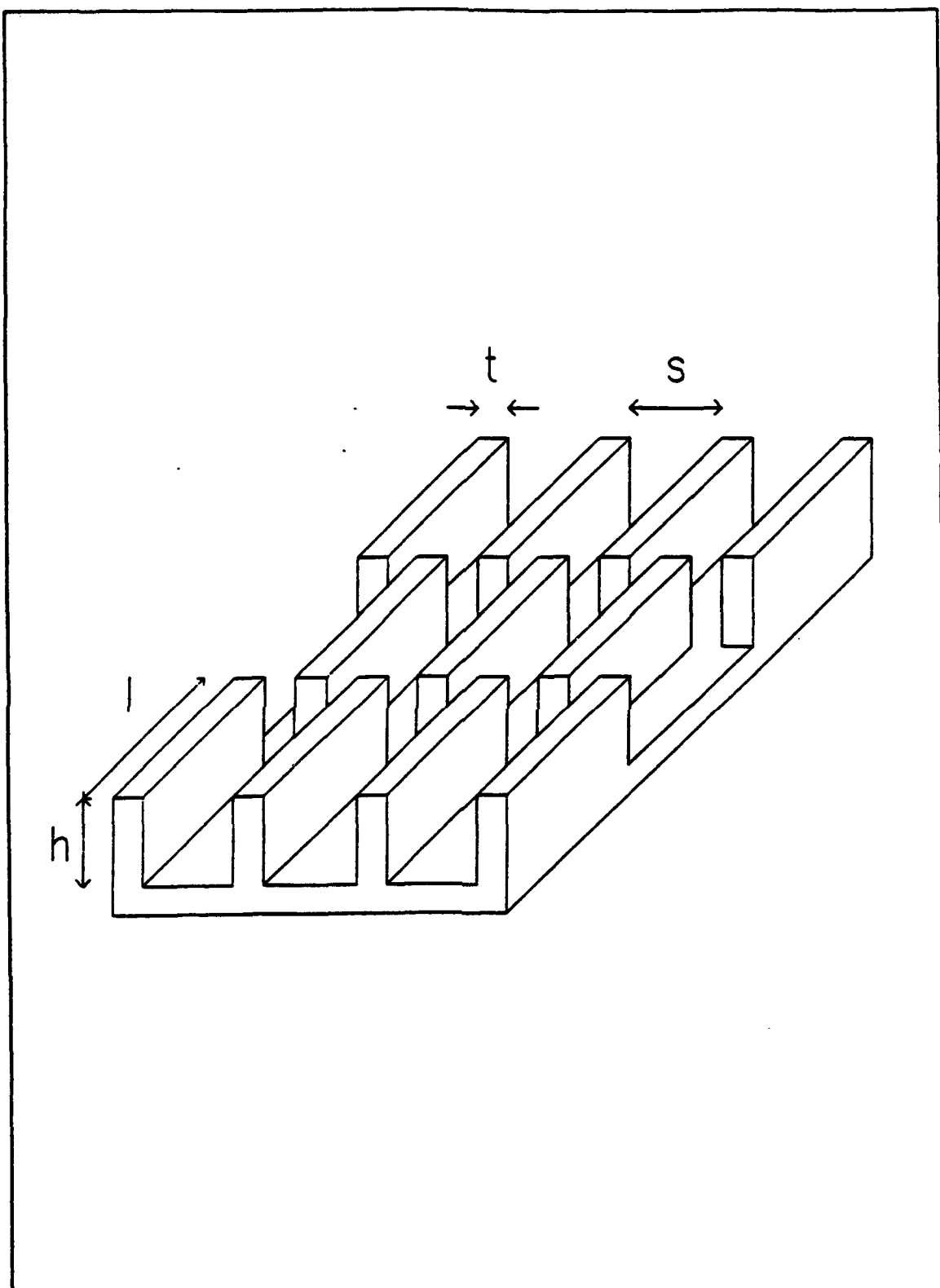


Figure 1. Offset Plate Fin Arrangement

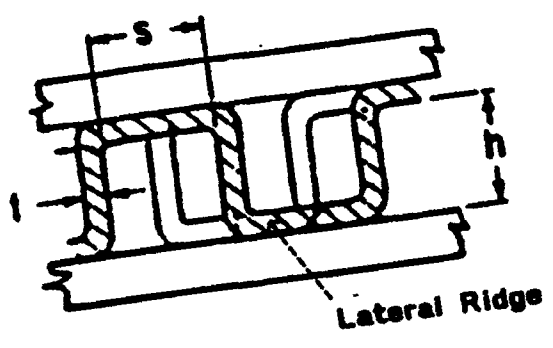
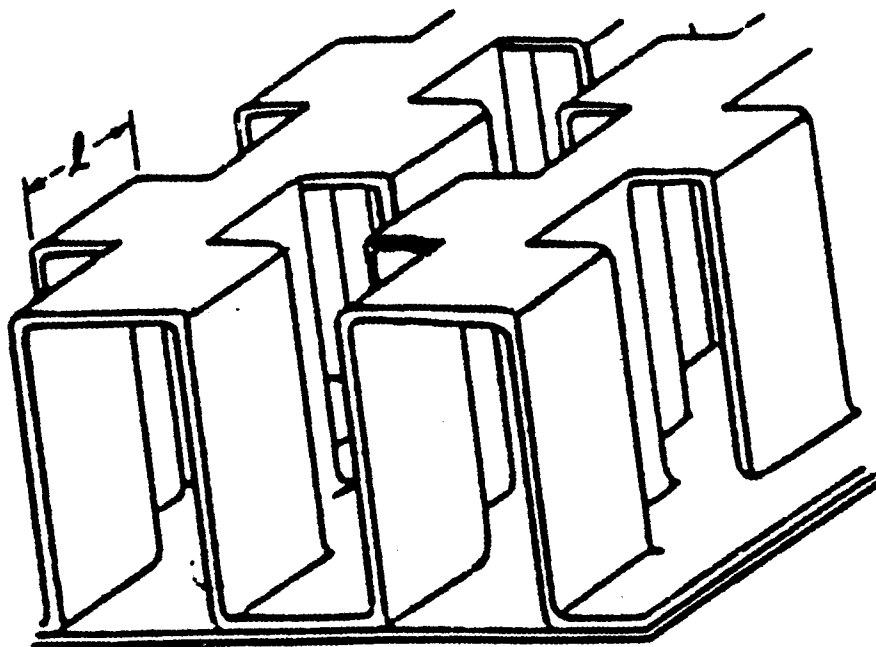


Figure 2. Offset Plate Fin Heat Exchanger Design (from Ref. 1)

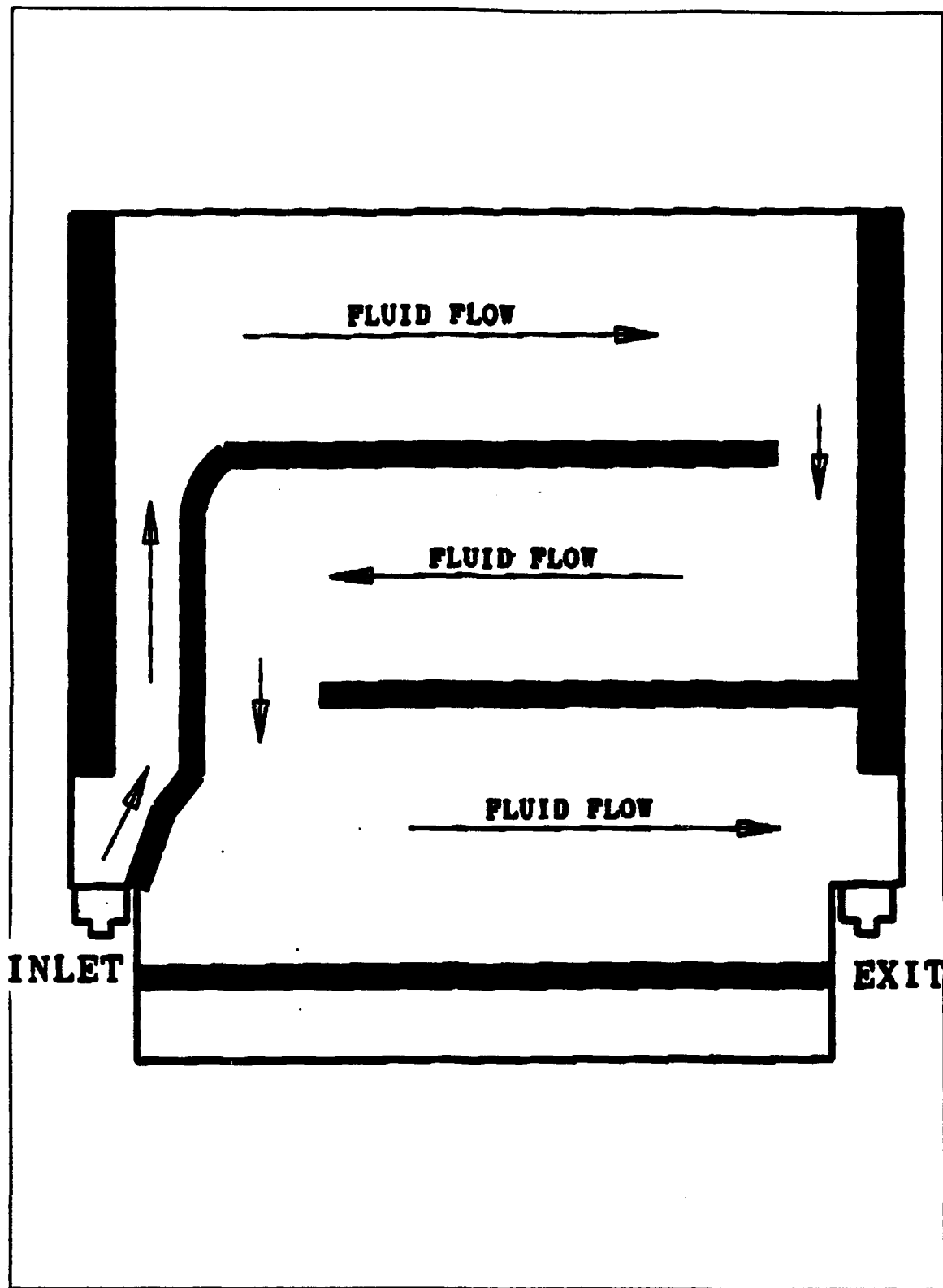


Figure 3. Liquid Flow Through Module (from Ref. 2)

II. EXPERIMENTAL APPARATUS

The experimental system consisted of the test section and the support system. A schematic of the entire system is shown in Figure 4. Descriptions of the sub systems and their components is provided next.

A. SUPPORT SYSTEM ASSEMBLY

1. Fluid Circulation System

An Endocal RTE-5 refrigerated circulating bath provided the constant temperature fluid surge volume to the inlet of the pump. The pumping power was provided by a Cole-Parmer positive displacement gear type pump driven by a variable speed motor producing Reynolds numbers in the range of interest (100 to 800). An Omega FTB-102 turbine flow meter and signal conditioner were used to measure fluid flow rates. The flow meter was calibrated by comparing the voltage output to quantities of water collected over specified times. The maximum uncertainty in measured flow rate was 0.5%.

Following the flow meter, the fluid temperature was measured using an Omega T-type thermocouple probe in the inlet tubing. The probe fed into the center of the tube through a T-fitting. A manifold was used to distribute the flow to three tubes that entered the test section. Three outlets from the test section returned the fluid to the circulating bath.

Interconnections between components were made with plastic and tygon tubing, and teflon fittings.

2. Power Distribution System

Power to the heater was supplied by a Kepco 0-100 V, 0-5 A power supply. The heater itself was a Minco 25.4 cm by 30.48 cm foil backed patch applied with pressure sensitive adhesive. It had an effective heating area of 1116 cm² and a total resistance of $11.2 \pm 1.1 \Omega$. The power supply was connected in series with a precision resistor ($R = 0.1 \pm 0.001 \Omega$) for heater current determination.

Two Keithley 169 digital multimeters were used to measure voltages. First, the voltage drop across the precision resistor (V_{pr}) was measured so that current to the heater could be accurately determined according to:

$$I_h = V_{pr} / 0.1 \Omega$$

The next measurement was the voltage drop across the heater itself (V_h). From these measurements, the power supplied to the heater could be calculated using the following relationship:

$$Pwr = V_h * I_h$$

3. Differential Pressure Measurement

Four pressure taps were located on the test section as indicated in Figure 4. The taps were provided with valves so two could be used at one time and the other two isolated.

Tygon tubing connected the pressure taps to a differential pressure transducer calibrated in the range of 0-254 mm H₂O.

4. Data Acquisition System

A HP 9000 computer was used to control a HP 3852 data acquisition unit for temperature and voltage measurements. There were 29 thermocouples on the test section and one inlet thermocouple that were scanned, with the output directed to the computer. In addition to the temperature measurements, the DC voltage output from the flow meter was read and converted to flow rate by the computer.

B. TEST SECTION ASSEMBLY

1. Fins and Base

Figure 5 shows a top view of the test section indicating the overall dimensions. Note that the part of the base covered by the plexiglass is not included in the dimensions as will be explained later in this section. The fins and the inlet and outlet plenum were made from aluminum alloy 6061. The fin dimensions were determined by scaling up the dimensions of a SEM-E fin by a factor of ten. This will be useful in generating a predictive model for the heat transfer performance of an SEM-E used in industrial applications.

To fabricate the fins, 2.54 cm (1") thick aluminum plate was cut into 31.75 mm wide blanks. These blanks were then milled to the proper length prior to cutting the fins

out. A specially made 11.76 mm diameter end mill was then used to mill the grooves in the blanks to form the fins with dimensions as shown in Figure 6. This left a 10.16 mm base on each fin section as indicated in Figure 7. One end of the blank was left longer than the other so that when alternated, each successive row of fins was offset by half the spacing (see Figure 7). An extra 12.7 mm (0.5") was also left on each end to accommodate the plexiglass cover. Prior to assembling the fins, all edges were filed by hand to ensure burr free surfaces.

The inlet and outlet plenums were made from 12.7 mm (0.5") aluminum plate. They had to be milled down to 10.16 mm to match the base of the finned sections. This prevented a ridge from disrupting the flow pattern upon entry into and exit from the fins.

Holes were drilled lengthwise through each fin section near the end to accommodate a threaded rod used to hold the 15 fin sections and the plenums together. Washers and nuts were used at each end of both rods and tightened appropriately for final assembly. Rubber insulating strips were placed between the first row of fins and the inlet plenum and similarly between the last row of fins and the outlet plenum. The purpose of the rubber strip was to minimize conduction heat losses from the finned section longitudinally to the plenums. Prior to final assembly, a bead of RTV (silicone rubber

coating) was placed between the fin sections to prevent leakage.

2. Plexiglass Cover

With the base assembled, the final measurements could be taken to build the cover to enclose the fins and plenums. It was made from 12.7 mm (0.5") thick plexiglass sheet. The sides were milled to 15.24 mm high (the same height as the fins) and cut to length. The top was then cut to be the same size as the base and the sides were cemented and screwed in. An acrylic cement was used to seal all the plexiglass components.

3. Inlets and Outlets

Three inlets and three outlets were placed at the ends as shown in Figure 4. These consisted of 3.175 mm (1/8") pipe thread drilled and tapped into the plexiglass wall with a 6.35 mm (1/4") tygon adapter on the opposite end of the fitting. Valves were placed on two of the outlets so they could be closed at low flow rates to provide a back pressure for filling and operating the test section as needed.

4. Pressure Taps

Four pressure taps were placed in the plexiglass cover to allow monitoring of various lengthwise pressure drops. The holes that penetrate into the test section were 1.588 mm (1/16") and were deburred by hand to ensure uniformity. On top of these holes, 3.175 mm (1/8") pipe taps were drilled and

threaded to accommodate the valves and tygon adapters. Valves were used since only two pressure taps were in service at a time.

5. Flow Straightening Region

In conjunction with the three inlets, a flow straightening region was used in the inlet plenum to provide a velocity profile as uniform as possible. The flow first hit a plastic mesh designed to distribute it evenly to the straws. The 38.1 mm ($1\frac{1}{2}$ ") straws were then used to develop a straightened flow prior to reaching the fins. The placement of these components is shown in Figure 5.

6. Final Assembly

With all the fixtures mounted, the test section could be assembled. The top was fitted over the fins and screwed through the aluminum into the plexiglass. RTV was again used as a sealant. A photograph of the completed test section is included in Figure 8. During operation, the test section was insulated with foam rubber to minimize any heat losses to ambient.

7. Test Section Instrumentation

Once assembled, Copper Constantan (T-type) cement-on thermocouples were mounted on the back of the finned section and the inlet and outlet plenums. The thermocouples used were flat ribbon type to allow better surface contact with the heater. Placement is indicated in Figure 5. The

thermocouples were mounted using a high thermal conductivity epoxy (Omegabond 101) along their leads. The junction of each sensor had a small amount of thermally conductive paste applied to minimize the thermal contact resistance.

After the thermocouple epoxy had cured, the heater was installed to the back of the test section. The entire finned part of the test section was covered except for 9.525 mm at each end and 3.24 mm on either side. A linoleum roller was used to apply sufficient pressure for good adhesion.

C. EXPERIMENTAL PROCEDURE

Both the Colburn j factor and the Fanning friction factor were desired as functions of Reynolds number. This required setting the flow rate as needed to obtain the Reynolds number of interest. With the flow held constant, the power to the heater was set to give a sufficient heat flux to allow a temperature gradient to be developed along the test section. After allowing time to ensure steady state, the necessary data was collected, reduced, and analyzed. This procedure was repeated for various flow rates, heat fluxes, and inlet temperatures until the entire range of Reynolds numbers obtainable with the available equipment had been covered.

Next, the test section was inclined at an angle of 45° (inlet lower) and an identical procedure as above was followed. This was to determine if there were any natural convection effects at low flow rates.

Lastly, a single temperature channel was monitored to determine if there were any time dependent temperature fluctuations within a single row of fins.

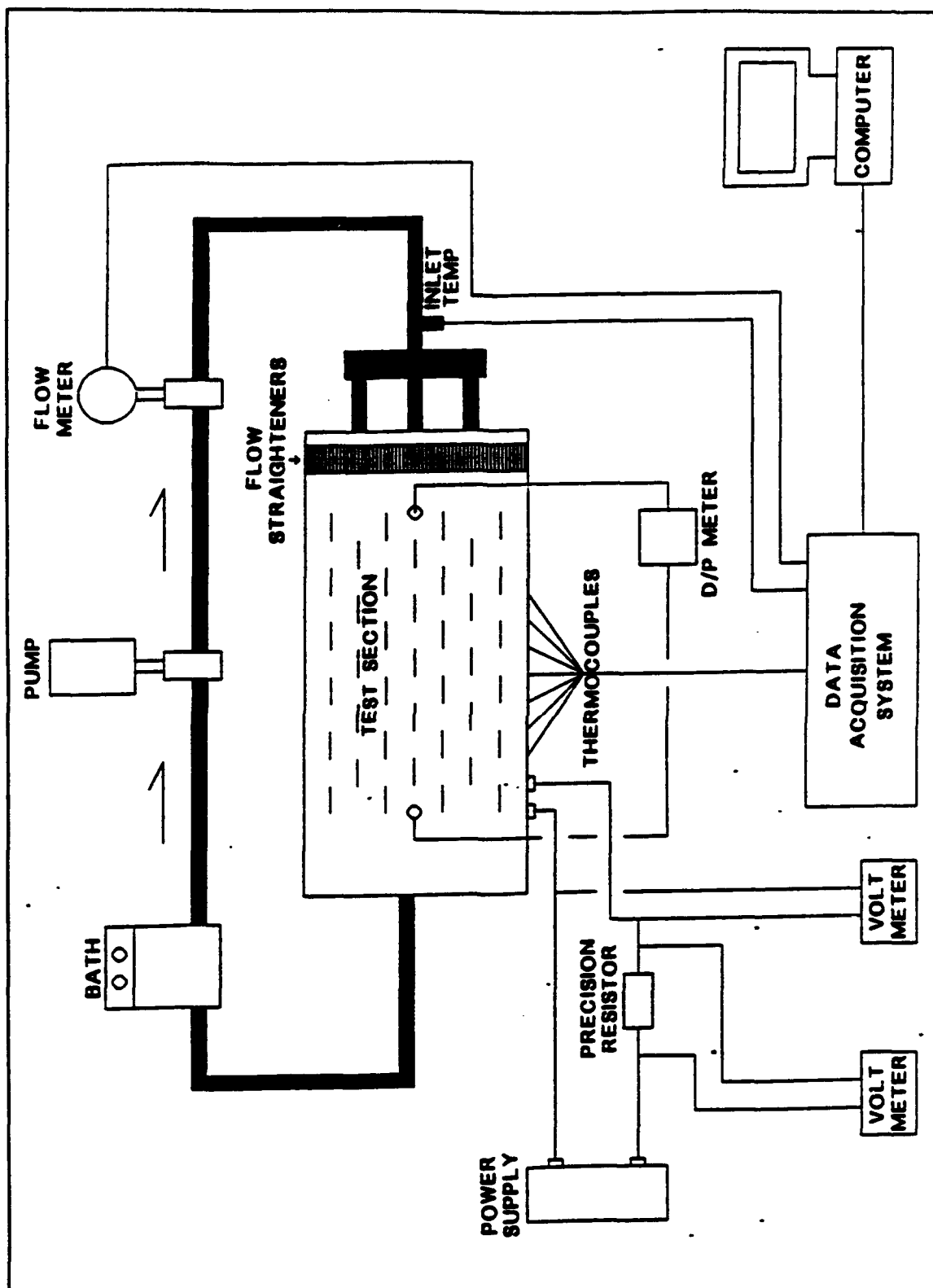


Figure 4. Overall Schematic of Experimental Apparatus

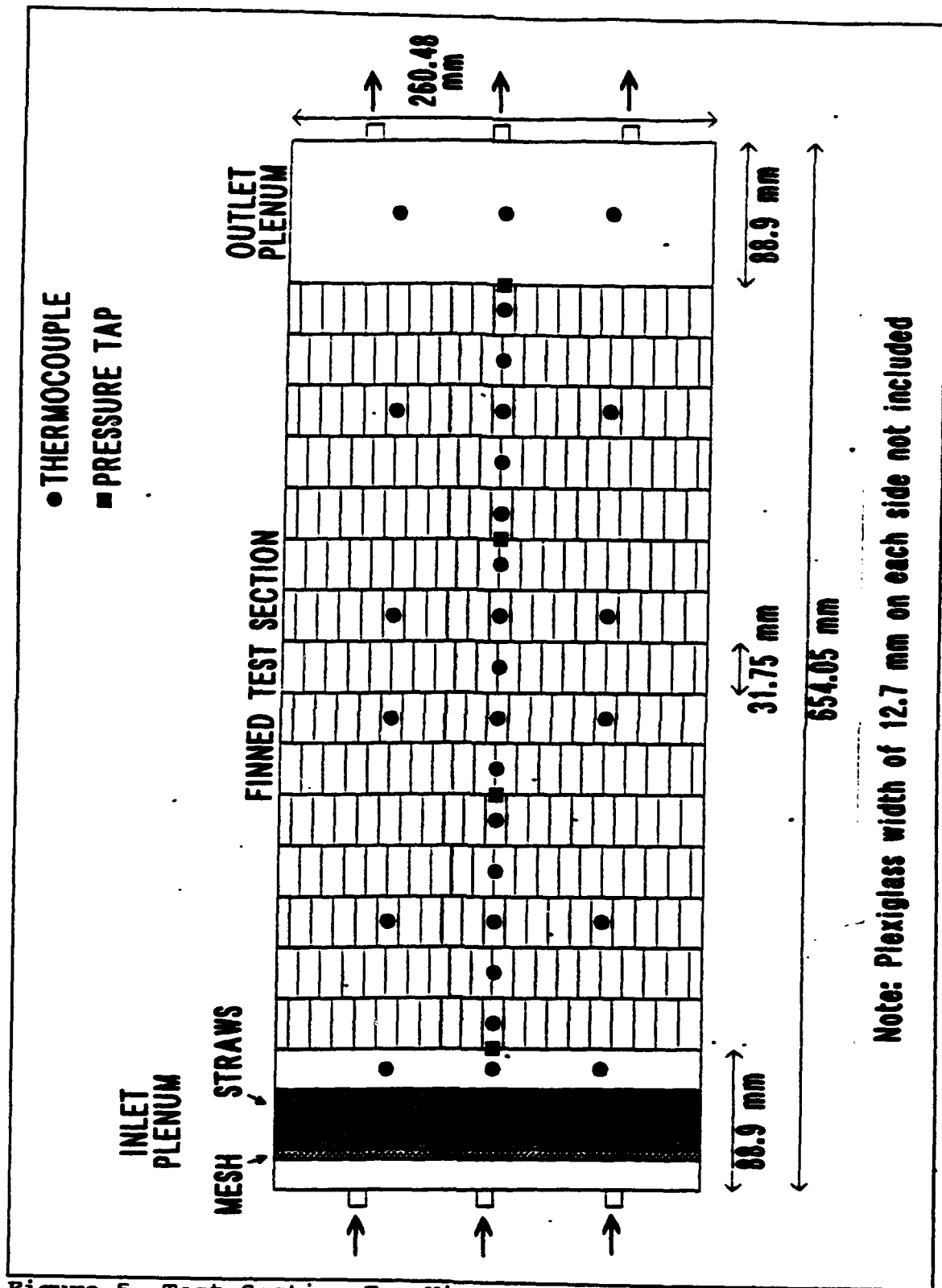


Figure 5. Test Section Top View

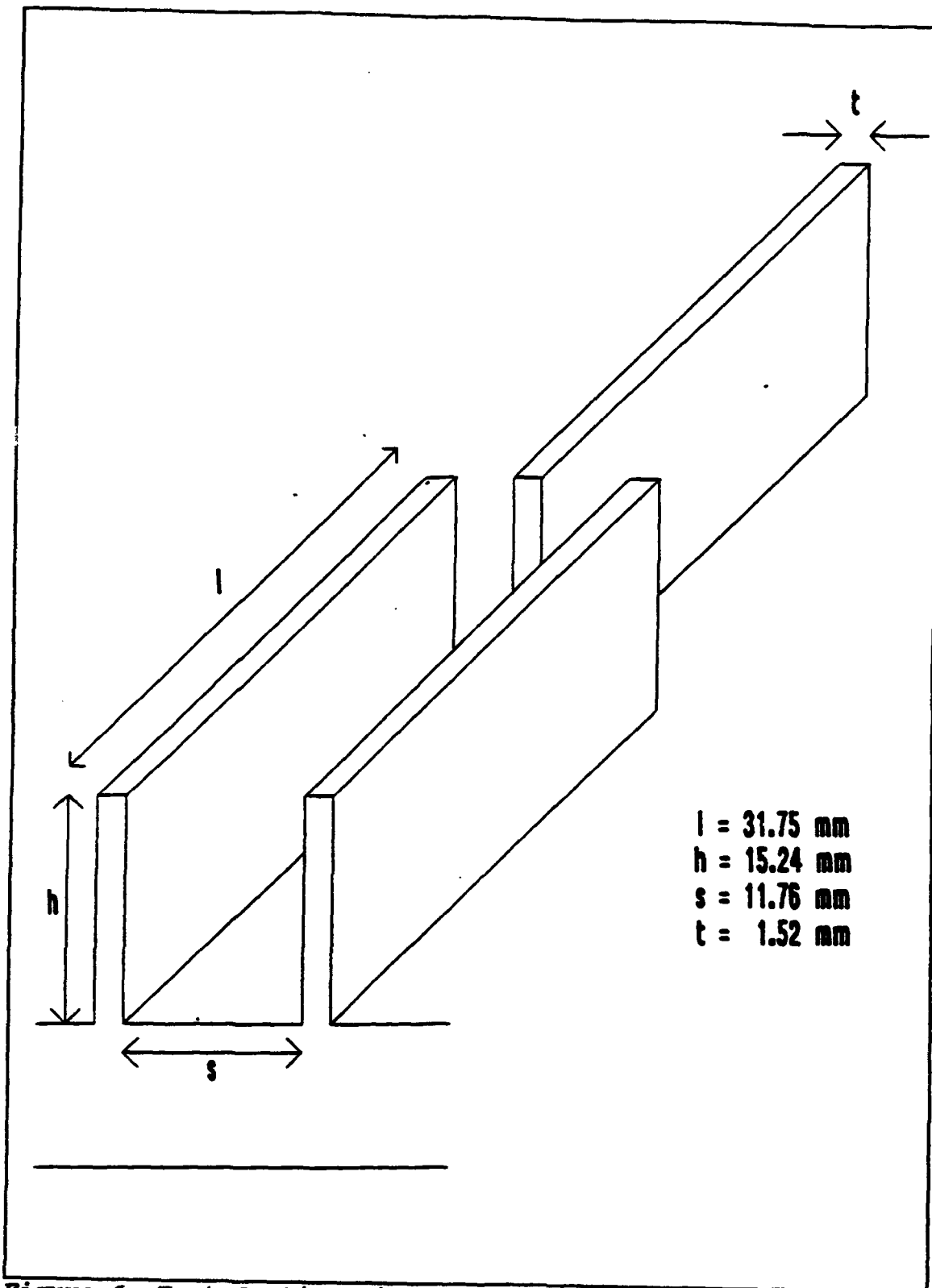


Figure 6. Test Section Fin Dimensions

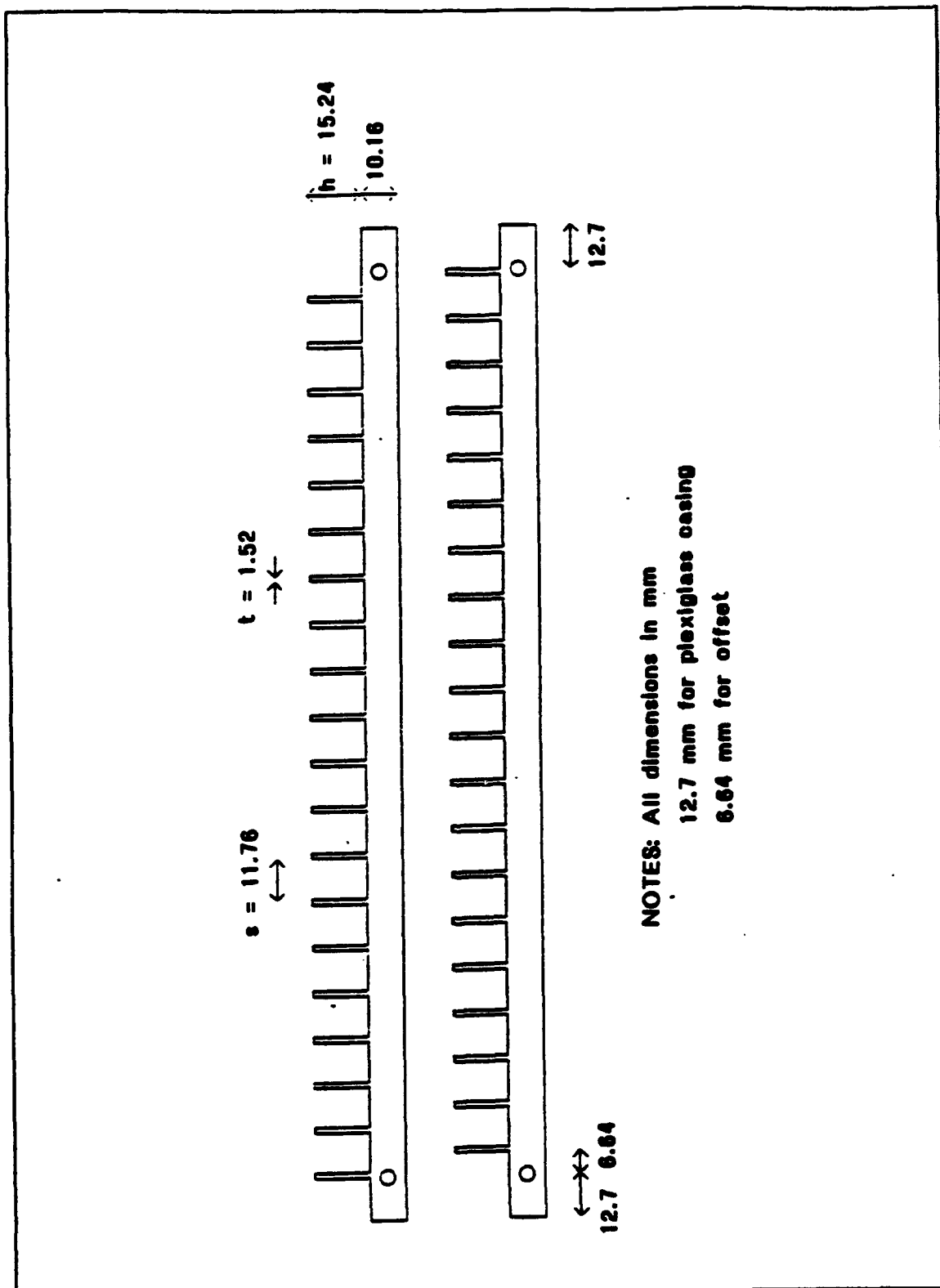


Figure 7. Finned Section Design

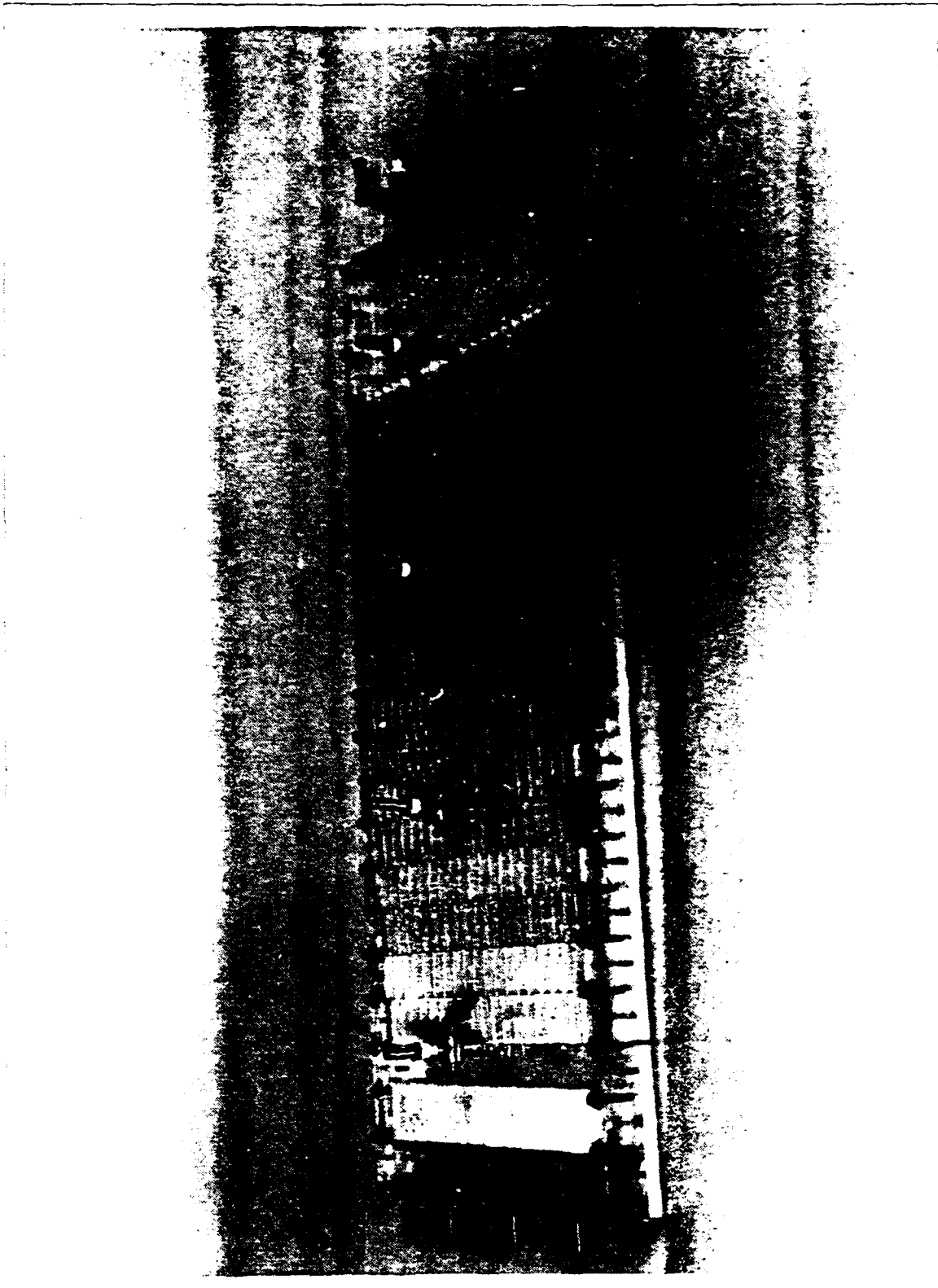


Figure 8. Completed Test Section

III. RESULTS

A. DATA ANALYSIS

Heat transfer data is plotted against Reynolds number which was calculated by:

$$Re = \frac{\rho * v * D_h}{\mu}$$

where D_h is defined by:

$$D_h = \frac{4 * A_c}{(A/l)}$$

In this case, A_c is the free flow area of each channel (=sh ; see Fig. 6) and (A/l) is the heat transfer area per unit channel length ($= [s_1+2h_1+2t_h]/l$). This is different than the definition for an actual offset plate fin heat exchanger since the plexiglass cover is not considered a heat transfer surface. The velocity through the channels was determined by:

$$v = \frac{Q}{A_f}$$

where Q is the volumetric flow rate from the pump and A_f is the free flow area through the test section (19 channels and 1 half channel).

The fluid properties, density (ρ), viscosity (μ), thermal conductivity (k_f), and Prandtl number (Pr), were all curve fitted in the range of 0-40 °C to aid in data reduction. The temperature used to evaluate the properties was determined according to:

$$T_{PROP} = \frac{T_{AVG} + T_{IN}}{2}$$

The Colburn j factor is typically used to describe the performance of offset plate fin heat exchangers. In order to calculate this, the average Nusselt number along the length of the heat exchanger is needed. It is defined by:

$$Nu_{AVG} = \frac{q'' * D_h}{k_f * (T_{AVG} - T_{IN})}$$

The heat flux (q''), is determined by:

$$q'' = \frac{P_{WIR}}{A_h}$$

where A_h is the effective area of the heater.

The inlet temperature to the test section (T_{IN}), was measured directly while the average temperature (T_{AVG}), was the average of the test section centerline thermocouples. No correction was made for conduction through the aluminum base

because it was estimated at less than 0.2 °C at the highest heat flux and therefore neglected.

With Nu_{Avg} known, the Colburn j factor could be calculated according to:

$$j = \frac{Nu_{Avg}}{Re * Pr^{1/3}}$$

B. HEAT TRANSFER RESULTS

The primary reason for investigating the heat transfer performance was to compare the results with published Colburn j factor correlations and to use the data to validate a future numerical model.

The results are tabulated in Tables 1 and 2 and shown graphically in Figure 9. The solid line in Figure 9 is a frequently cited correlation developed by Wieting [Ref. 4] using air as the working fluid in the laminar flow range ($Re < 1000$). It is given by:

$$j = 0.483 * \left(\frac{1}{D_h} \right)^{-0.162} * \left(\frac{s}{h} \right)^{-0.184} * (Re)^{-0.536}$$

It should be noted that Wieting defines D_h differently ($D_h = 2sh/[s+h]$) than the present study (see Chapter III, Section A) but the plotted correlation was corrected by using the fin

dimensions from this study with Wieting's definition of hydraulic diameter.

The dashed line in Figure 9 is a newer correlation developed by Manglik and Bergles [Ref. 3] using data gathered since Wieting's study. They state that their correlation is valid for a Prandtl number range of 0.5-15. This would include water at normal temperatures but would not include high Prandtl number heat transfer oils such as PAO. The correlation is given by:

$$j = 0.6522 * (Re)^{-0.5403} * \left(\frac{s}{h}\right)^{-0.1541} * \left(\frac{t}{l}\right)^{0.1499} * \left(\frac{t}{s}\right)^{-0.0678}$$

Note that this correlation is based solely on Reynolds number and fin dimensions so does not require a correction for D_h .

The data was also compared to the preliminary results of the Maryland study [Ref. 1]. While their data is not shown in Figure 9, there was very close agreement with the results of this study.

1. Horizontal Test Section

Fifteen data runs were performed with the test section in the horizontal position. The Colburn j factor data points are shown as asterisks in Figure 9. They compare favorably with the previous correlations but tend to deviate slightly at higher Reynolds numbers. This could possibly be due to nearing the transition range to turbulent flow. This range is

defined by Wieting to be between Reynolds numbers of 1000-2000 [Ref. 4].

TABLE I. HORIZONTAL TEST SECTION DATA

Re	q'' (W/m ²)	ΔT (°C)	Nu_{avg}	j
211	2485	6.0	11.2	0.027
224	2025	6.4	8.7	0.020
245	3495	9.7	9.8	0.021
263	2485	6.0	11.3	0.022
295	1960	5.2	10.2	0.018
305	2485	5.8	11.6	0.020
322	3495	9.6	9.8	0.016
380	3505	7.9	11.9	0.017
441	3505	7.8	12.1	0.015
518	3510	7.5	12.6	0.013
563	3505	7.4	12.7	0.012
623	3505	7.3	12.8	0.011
674	3505	7.6	12.4	0.010
731	3515	8.0	11.7	0.009
770	3515	6.9	13.6	0.010

It should be noted that there were considerable variations in temperature between the monitored thermocouple locations. The reason for this is not entirely clear but believed to be due to the complex flow paths through the fins. The variations occurred both lengthwise and spanwise through the test section. Attempts were made to determine whether or not this was a measurement anomaly. Each test indicated that the measured temperatures were correct and they stayed consistent throughout the duration of the experiment.

2. Inclined Test Section

Five additional data runs were performed with the test section inclined at an angle of 45° in order to study the effects of natural convection at low flow rates. The resulting Colburn j factor data points are plotted as circles in Figure 9. They tend to fall in line with the data points from the horizontal runs.

C. Time Dependent Temperature Variation

With the test section at steady state at Reynolds numbers between 400-600, a single channel was monitored at a sampling rate of 2.5 samples per second to attempt to detect any time dependent behavior. Each row of fins was checked but none of them varied by more than 0.1°C .

TABLE II. INCLINED TEST SECTION DATA

Re	q'' (W/m ²)	ΔT (°C)	Nu_{avg}	j
141	1590	4.8	8.9	0.032
193	1590	4.6	9.3	0.025
218	2025	5.5	10.1	0.023
271	2605	6.7	10.6	0.020
367	3505	8.5	11.1	0.016

D. PRESSURE DROP RESULTS

It was not possible to measure the pressure drop through the test section so no Fanning friction factor data is included. Attempts were made with two differential pressure measurement systems but the pressure drop proved too small to measure with the equipment available. It was determined that it was on the order of a few mm H₂O.

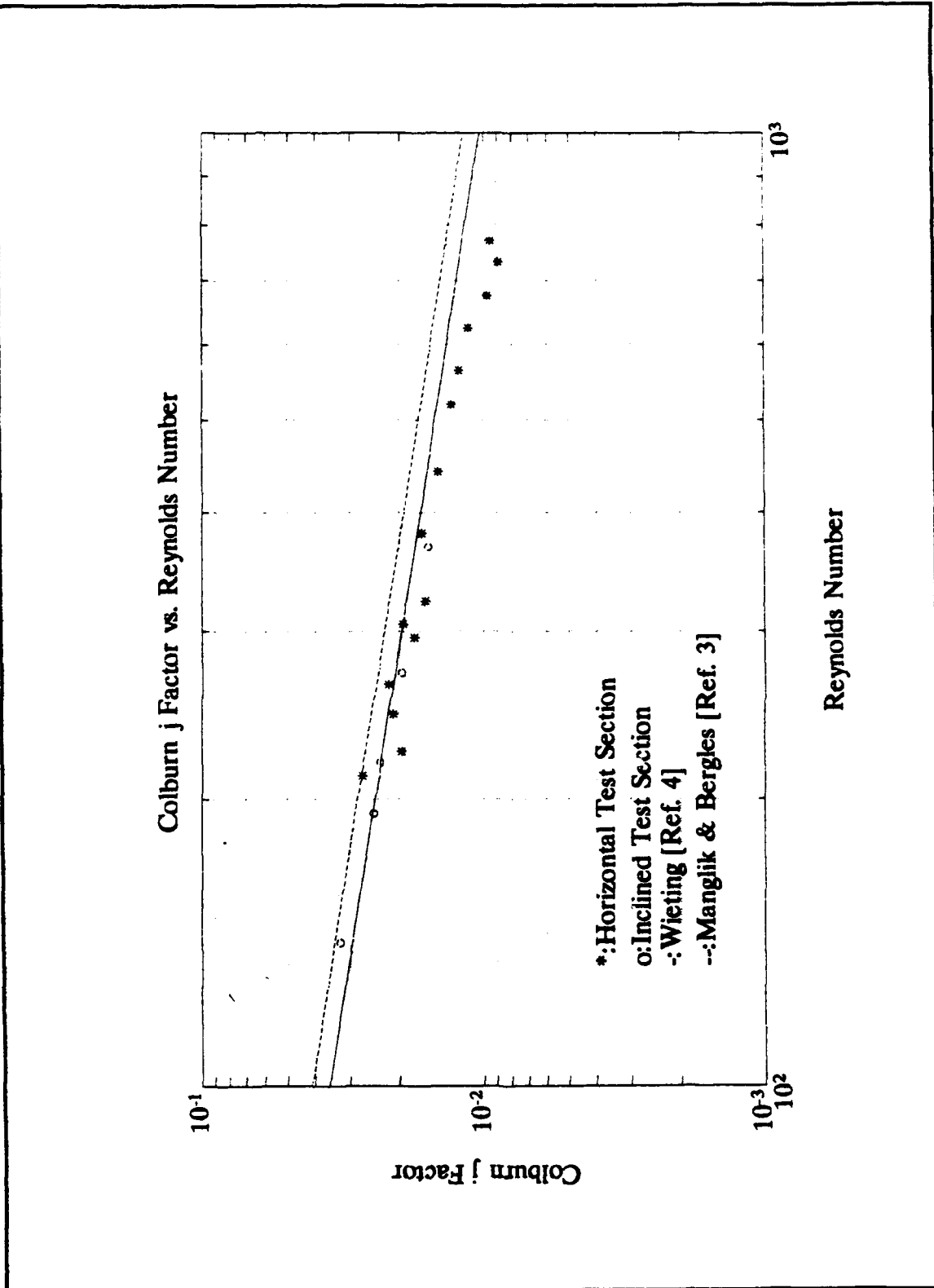


Figure 9. Re vs j for Test Section

IV. CONCLUSIONS

This study reached the following conclusions on the heat transfer performance of an offset plate fin heat exchanger.

The Colburn j factor data agreed favorably with previous correlations which tends to validate the experiment. The differences between how the measurements were made in this study as opposed to previous studies also tends to validate the correlations.

Previous studies used actual heat exchanger cores, vice scaled up models, and measured an average surface temperature of the core to use in their calculations. This study was the first attempt, to the author's knowledge, to measure the surface temperature of the individual rows of fins on a scaled up model. This technique provided some insight into the complexities of the flow within the finned region and allowed an average temperature to be calculated based on each row of fins. This is not possible on actual cores due to the small size of the fins.

The inclined heat transfer data showed that for Reynolds numbers as low as 140, natural convection does not appear to play any significant role in the heat exchangers performance.

The agreement of the Colburn j factor data with the correlations for air indicates that the Prandtl number for water is not high enough to give any enhancement in heat

transfer. In order to get a significant increase in performance, higher Prandtl number fluids must be used.

In general, the results of this study, when combined with the results of future studies, will allow researchers to validate numerical models in order to predict offset plate fin heat exchanger performance. These numerical models will then hopefully be extended so that they can model heat exchangers using higher Prandtl number fluids.

V. RECOMMENDATIONS

In continuation of this study, it is recommended that follow on studies include:

- Pressure drop data in order to determine the Fanning friction factor.
- Flow visualization using either liquid crystals or dye injection.
- The use of high Prandtl number fluids such as PAO in order to aid in the determination of the Prandtl number dependence on heat transfer performance.

APPENDIX A

UNCERTAINTY ANALYSIS

An uncertainty analysis was performed to evaluate the accuracy of the data in this study.

The uncertainty of a function $F = F(X_1, X_2, X_3)$, where $X_{1,2,3}$ are the independent measurements is determined by:

$$\delta F = \left[\left(\frac{\partial F}{\partial X_1} \delta X_1 \right)^2 + \left(\frac{\partial F}{\partial X_2} \delta X_2 \right)^2 + \left(\frac{\partial F}{\partial X_3} \delta X_3 \right)^2 \right]^{1/2}$$

For $F = C X_1^a X_2^b X_3^c$ the uncertainty can be put in the following form:

$$\frac{\delta F}{F} = \left[\left(a \frac{\delta X_1}{X_1} \right)^2 + \left(b \frac{\delta X_2}{X_2} \right)^2 + \left(c \frac{\delta X_3}{X_3} \right)^2 \right]^{1/2}$$

1. Reynolds Number Uncertainty

$$Re = \frac{\rho * v * D_h}{\mu}$$

and

$$\frac{\delta Re}{Re} = \left[\left(\frac{\delta v}{v} \right)^2 + \left(\frac{\delta D_h}{D_h} \right)^2 \right]^{1/2}$$

where,

$$\frac{\delta D_h}{D_h} = \left[\left(\frac{\delta A_c}{A_c} \right)^2 + \left(\frac{\delta (A/l)}{(A/l)} \right)^2 \right]^{1/2}$$

For

$$A_c = 0.000179 \text{ m}^2$$

$$\delta A_c = 0.000002 \text{ m}^2$$

$$(A/l) = 0.0437 \text{ m}$$

$$\delta (A/l) = 0.00025 \text{ m}$$

$$v = 0.0218 \text{ m/s}$$

$$\delta v = 0.0003 \text{ m/s}$$

$$D_h = 0.0164 \text{ m}$$

$$\delta D_h = 0.00096 \text{ m}$$

$$\rho = 997.7 \text{ kg/m}^3$$

$$\mu = 0.0009374 \text{ N-s/m}^2$$

$$Re = 380 \pm 23$$

The uncertainty $\delta Re/Re = 0.060$ or 6.0%

2. Colburn j Factor Uncertainty

$$j = \frac{Nu_{avg}}{Re * Pr^{1/3}}$$

and

$$\frac{\delta j}{j} = \left[\left(\frac{\delta Nu}{Nu} \right)^2 + \left(\frac{\delta Re}{Re} \right)^2 \right]^{1/2}$$

where,

$$\frac{\delta Nu}{Nu} = \left[\left(\frac{\delta q''}{q''} \right)^2 + \left(\frac{\delta D_h}{D_h} \right)^2 + \left(\frac{\delta \Delta T}{\Delta T} \right)^2 \right]^{1/2}$$

For

$$q'' = 3505 \text{ W/m}^2$$

$$\delta q'' = 36 \text{ W/m}^2$$

$$D_h = 0.0164 \text{ m}$$

$$\delta D_h = .00096 \text{ m}$$

$$\Delta T = 7.9 \text{ }^\circ\text{C}$$

$$\delta \Delta T = 0.1 \text{ }^\circ\text{C}$$

$$Nu = 11.9$$

$$\delta Nu = 0.7$$

$$Re = 381$$

$$\delta Re = 22.8$$

$$Pr = 6.458$$

$$j = 0.017 \pm 0.001$$

The uncertainty $\delta j/j = 0.085$ or 8.5%

APPENDIX B

SAMPLE CALCULATIONS

The following calculation is for $Q=7.63E-5$ m³/s, $q''=3505$ W/m², $T_{AVG}=300.1$ K, $T_{IN}=292.2$ K.

1. Characteristic Dimension

$$\begin{aligned}\text{Hydraulic Diameter } (D_h) &= (4)(1.79E-4)/(4.37E-2) \\ &= 0.0164 \text{ m}\end{aligned}$$

2. Water Properties [Ref. 5, p. A22]

$$T_{PROP} = (T_{AVG} + T_{IN})/2 = 296.1 \text{ K}$$

$$\rho = 997.7 \text{ kg/m}^3$$

$$\mu = 9.374E-4 \text{ N-s/m}^2$$

$$k_f = 0.6076 \text{ W/m-K}$$

$$Pr = 6.458$$

3. Reynolds Number

$$v = (7.63E-5)/(3.506E-3) = 0.0218 \text{ m/s}$$

$$Re = (997.7)(0.0218)(0.0164)/(9.374E-4) = 380$$

4. Nusselt Number

$$\begin{aligned}Nu_{AVG} &= (3505)(0.0164)/(0.6076)(300.1-292.2) \\ &= 11.9\end{aligned}$$

5. Colburn j Factor

$$j = (11.9)/(380)(6.458)^{1/3} = 0.017$$

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